II. Amendments to the Specification

Please amend the "Brief Description of Several Views of the Drawings" section of the specification as follows:

BRIEF DESCRIPTION OF SEVERAL VIEWS OF THE DRAWINGS

Fig. 1 is a perspective view of a prior art axle pinion gear having a splined interface.

Fig. 2 is a perspective view of an axle pinion gear having a male polygonal interface according to the present invention.

Fig. 3 is a perspective view of a companion flange having a female polygonal interface to match the interface of Fig. 2.

Fig. 4 is a perspective view of the embodiments of Figs. 2 and 3 assembled.

Fig. 5 is an end, perspective view of the axle pinion gear shaft with a polygonal interface.

Fig. 6 is a representation of the measurement of convexity or concavity of the polygonal surfaces.

Figs. 7 and 8 are polygonal surfaces according to the present invention.

Fig. 9 is a flow diagram illustrating a method of manufacturing an axle pinion gear having a twisted polygonal interface.

<u>Fig. 10 is a break away view of a portion of the internal interface portion of Figure 4.</u>

Please amend as follows the following consecutive paragraphs that begin on page 7, line 19 and end at page 11, line 2 of the specification:

Fig. 2 is a perspective view of an axle pinion gear assembly 24 according to the present invention. Axle pinion gear assembly 24 is meant for use in automotive applications, such as in trucks and automobiles, although other applications may also take advantage of the present invention. The axle pinion gear assembly 24 includes a gear 26 at one end for interfacing with a differential. The axle pinion gear assembly also includes a threaded surface 28 at the opposite end for a nut that will secure a companion flange in an axial direction. The axle pinion gear includes a polygonal interface ~~surface~~ 30, described below, in this case a hexagonal surface with a slight concavity on each of the six surfaces. The polygonal interface secures the companion flange in a radial direction. The axle pinion gear also includes journals 32 and 34 for bearing surfaces. Fig. 3 depicts matching companion flange 38 for the axle pinion gear assembly 24. Companion flange 38 preferably has an outer surface with a plurality of holes 42 for attachment to a drive shaft yoke, and also has a polygonal interface ~~surface~~ 40 to match the polygonal interface ~~surface~~ 30 of the axle pinion gear. The polygonal interface ~~surface~~ 40 of the companion flange has a slight convexity to match the concave surfaces of the axle pinion gear.

The assembled parts are depicted in Fig. 4. Axle pinion gear 24 and its polygonal interface ~~surface~~ 30 fit into companion flange 38 and its matching polygonal interface ~~surface~~ 40. The holes of the flange are available for mounting to a drive shaft yoke (not shown) and the threads 28 of the axle pinion gear are adapted to receive a retaining nut (not shown). Fig. 5 is an end, perspective view of a portion of the axle pinion gear of Fig. 2. As mentioned above, the axle pinion gear

comprises a threaded end 28, polygonal interface 30 ~~surfaces 30a, 30b, 30c~~, and at least one bearing surface 32. Polygonal interface 30 ~~surfaces 30a, 30b, 30c~~ may actually be separated into three portions as shown along the axis 25 of the shaft. The portions are preferably of roughly equal length, although this is not required, as will be seen.

First segment ~~portion~~ 30a is machined in alignment with the axis 25 of the axle pinion gear 24. Twisted segment ~~Portion~~ 30b is machined so as to provide a small twist, either clockwise or counterclockwise, relative to the axis of the shaft. Finally, second segment ~~portion~~ 30c is machined ~~with a second twist equal and opposite to that given to section 30b~~ in alignment with the axis 25 of the axle pinion gear 24. The effect of the middle segment ~~portion~~, 30b, is as though it were twisted along its outer surface. The angle is small, preferably from about 0° 10' to about 1°. In another embodiment, the angle is selected from a narrower range, from about 0° 20' to about 0° 50', and in yet another embodiment, the angle is close to about 0° 35'. It has been found that in shafts from about 1" diameter to about 3" in diameter, this twist in the middle section is effective in eliminating backlash. At the same time, the angle is not so great that it is difficult to assemble the parts using known methods for assembling parts with interferences. These methods include thermal techniques and techniques using a mechanical advantage.

The twist is only machined onto one of the two parts, preferably the male portion, while the matching part, for instance the female portion, is kept straight. It may be easier to machine the twist onto the male portion of the polygonal interface, that is, onto the shaft, although the twist may instead be machined onto the female portion. When assembled, the second segment ~~outer part~~, 30c fits readily into the

female portion. When further assembled, the twisted segment ~~portion~~ 30b induces torsion into the shaft and into the mating portion of the female polygonal interface. When the assembly is completed, first segment ~~portion~~ 30a will be in torsion against the mating portion of the female in one direction, and second segment ~~outer portion~~ 30c will be in torsion in the opposite direction~~, resisting the torsion of the middle portion~~. When the angle is kept small, these small interferences will eliminate backlash and thus reduce the wear of the male and female polygonal surfaces. An important part of the design of the polygonal interfaces is the interface itself and the degree of convexity or matching concavity. A polygonal surface according to the present invention may have from 3 to any number of sides. However, as the number of sides increases, manufacturing and programming complexity will also increase for programming the lathes that may be used to turn the shaft and manufacture the part. It has been found that polygonal parts with a relative eccentricity of up to about 4% may preferably be used. Eccentricity is defined as shown in Fig. 6. A polygonal (in this case, hexagonal) surface 44 is circumscribed by circle 46 at its outermost points. An inner circle 48 is scribed at the innermost points. The eccentricity (e) of the polygon is defined as the difference between the diameter of the outer circle 46 (D_{out}) and the inner circle 48 (D_{in}). [$e=1/2\ (D_{out} - D_{in})$]. The relative eccentricity (E) is defined in percentage terms as the eccentricity divided by the average diameter of the outer circle 46 (D_{out}) and the inner circle 48 (D_{in}). [$E = (e/\ D_{middle}) \times 100\%$], and [$D_{middle}=1/2\ (D_{out} + D_{in})$]. It is clear that as the inner circle approaches the outer circle, there is less eccentricity, until the sides of the "polygon" converge to a single circle ($e = 0$). While this certainly possible, it is preferable to have at least about 1.5% relative eccentricity in the concavity or convexity of the polygon used for

mating surfaces. The reason is that with a smaller eccentricity the tangential stresses tend to point closer to the center of the shaft, which in turn creates a higher shear stress. And with a greater eccentricity, especially concavity in a male driven member, the tangential stress points away from the center of the shaft, hence creating a lower shear stress. Therefore, while a concave surface on a male driven member is only one embodiment, it is a preferred embodiment.